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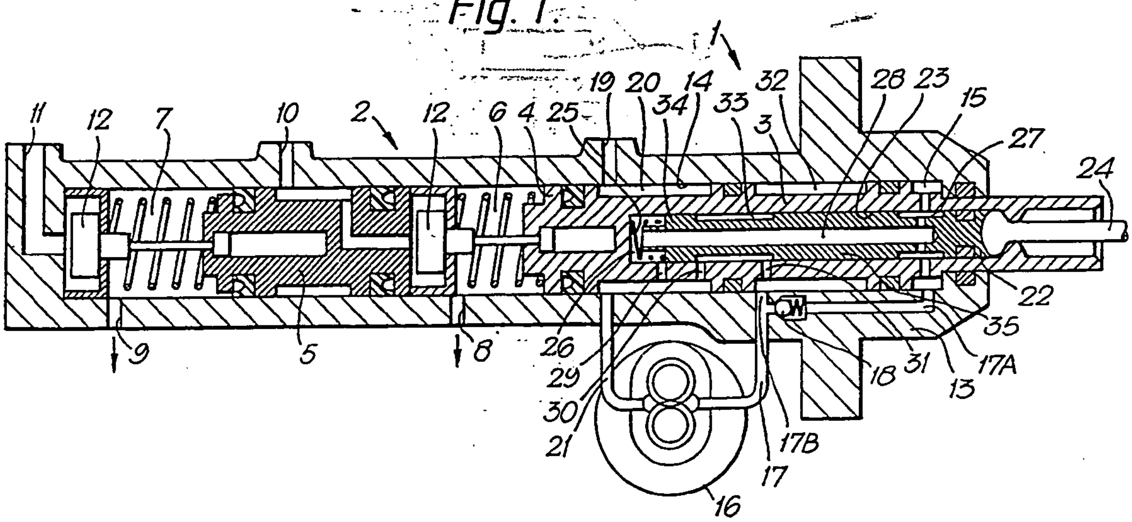
(54) **Servo for hydraulic master cylinder**

(57) A servo 1 for a hydraulic master cylinder 2 comprises: a body 13 defining a bore 14; a servo piston 3 mounted in the bore for providing an output force to a piston 4 of the master cylinder; a servo chamber 15 defined between the bore and the servo piston for receiving pressurized fluid to drive the servo piston along the bore; a pump 16 for supplying pressurized working fluid to the servo chamber; an input member 24 for receiving an input force from an operating pedal; and control means 31, 35 operative when the pressure in the servo chamber reaches a value determined by the magnitude of the input force to relieve the pump load whilst maintaining the pressure in the servo chamber at the desired level.

An alternative servo (102, Fig. 2) can be utilised for traction control. An electronically controlled solenoid valve (206, Fig. 3), in a third embodiment, is actuated in response to forces at load cells (216, 218) and vehicle loading.

Fig. 1.



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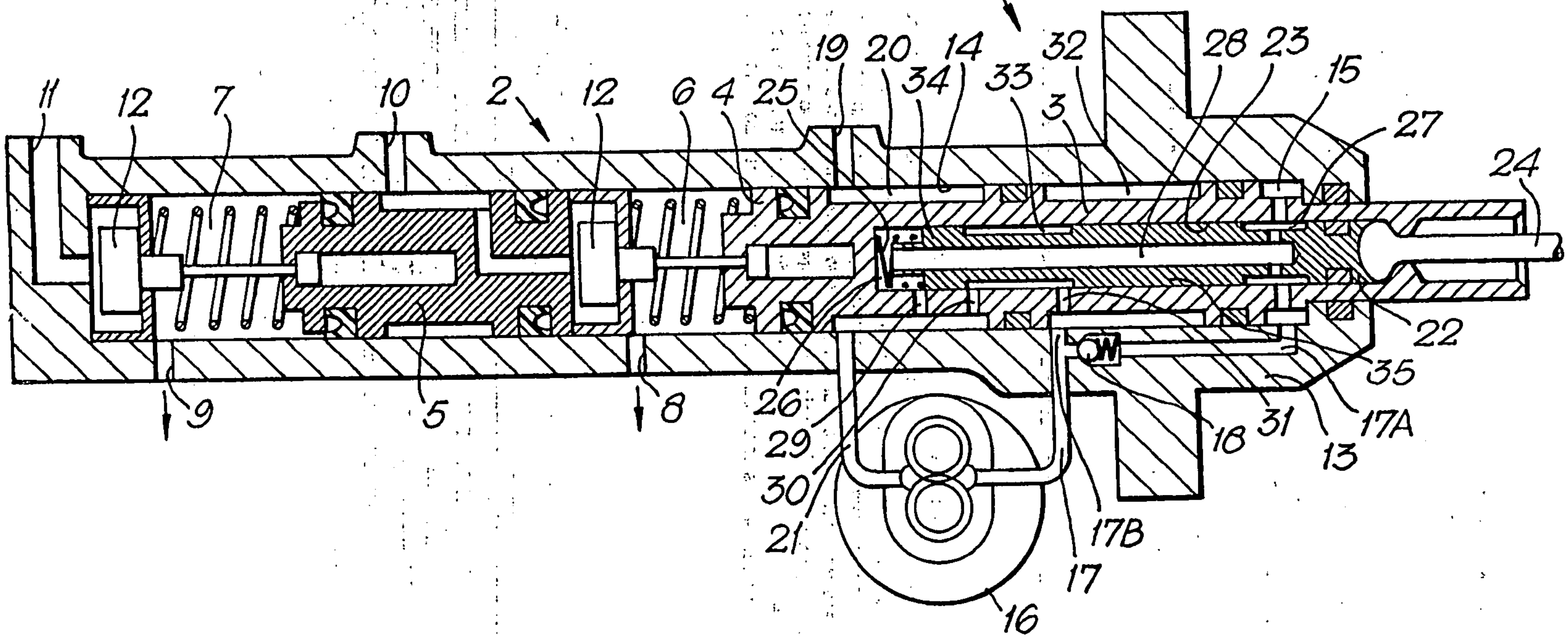
At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

The claims were filed later than the filing date within the period prescribed by Rule 25(1) of the Patents Rules 1990.

Fig. 1.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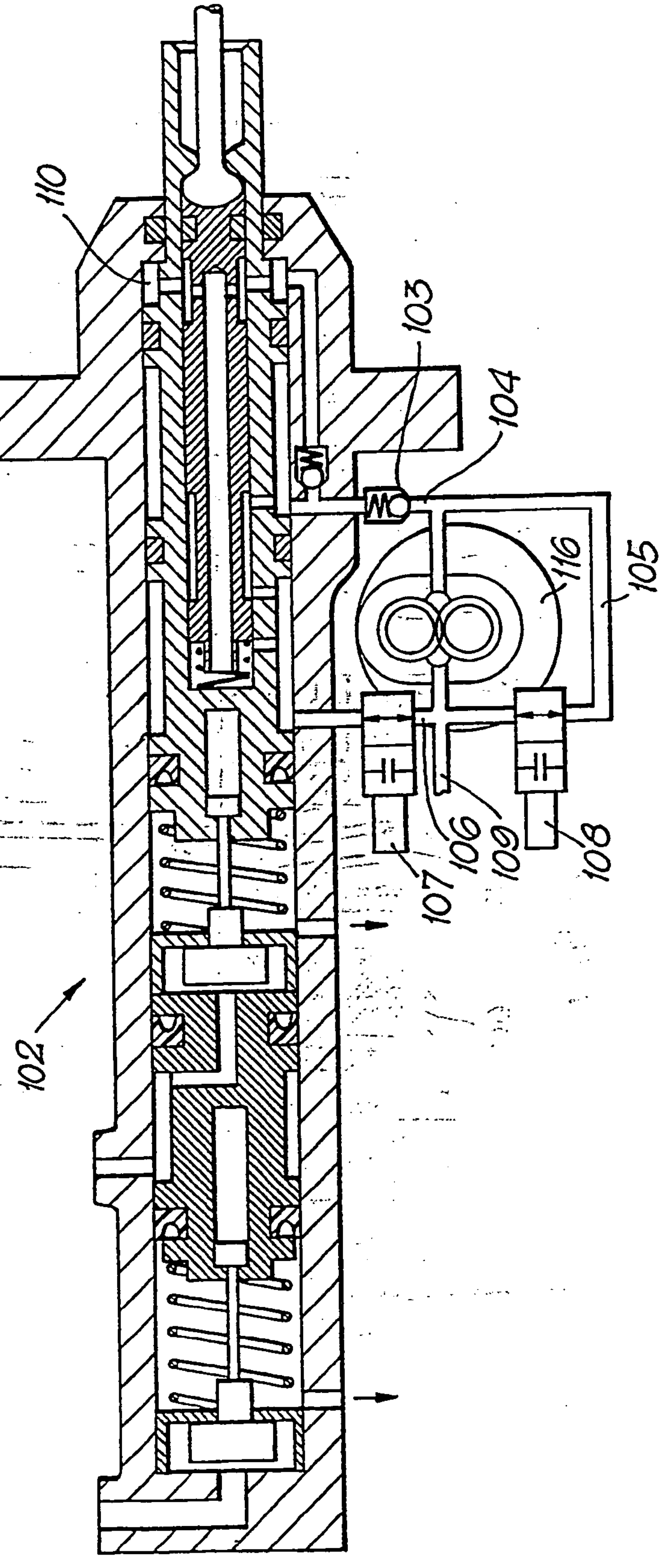

Fig. 2.

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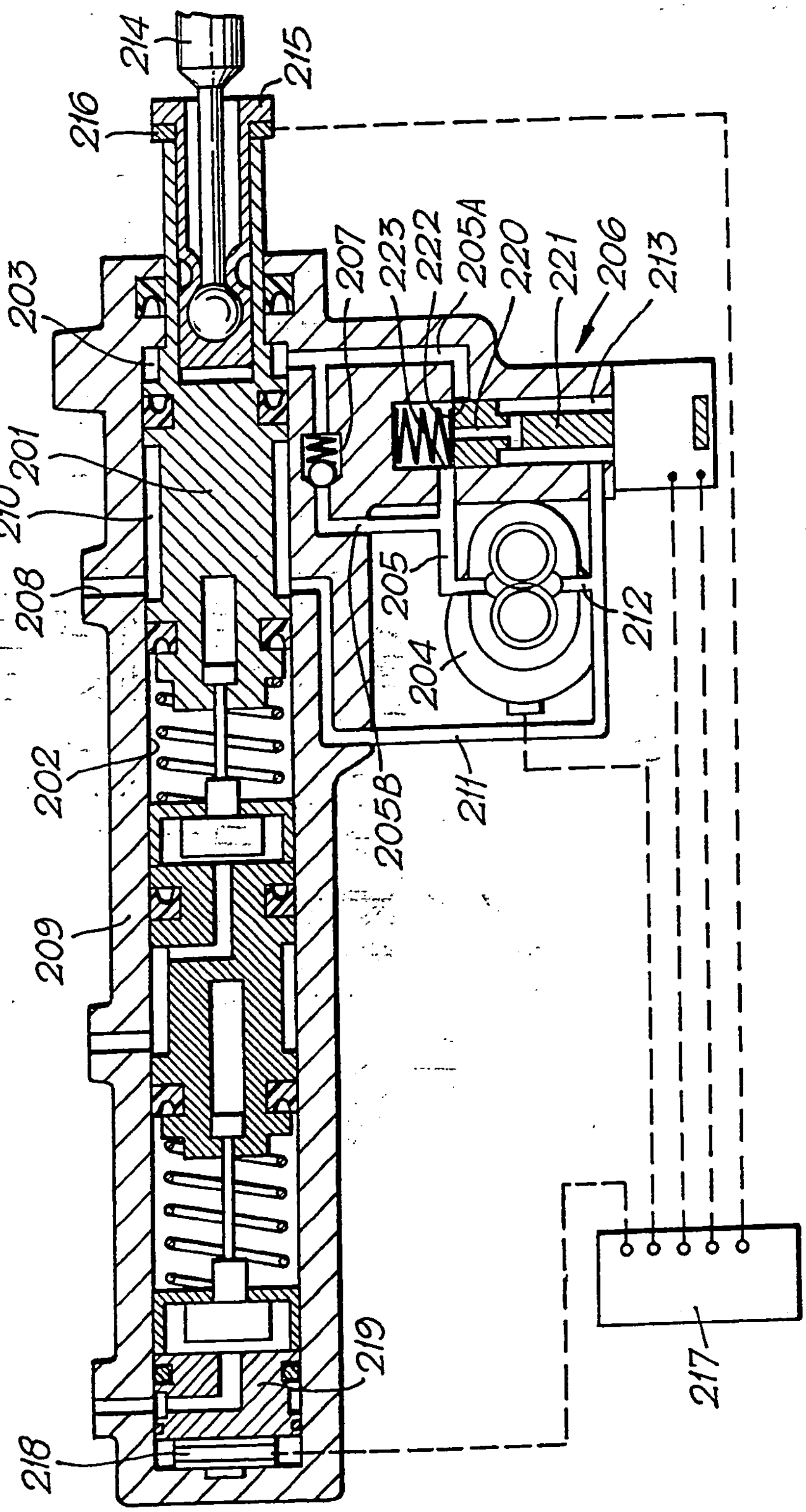

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205B
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218
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Fig.3.

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SERVO FOR HYDRAULIC MASTER CYLINDER

This invention relates to a servo for a hydraulic master cylinder, that is to say a device which utilizes a suitable power source to provide an input force to the hydraulic master cylinder which is larger than the force applied to the servo by the user. Such servos are very commonly used in vehicle braking systems.

Heretofore, the most common form of servo in use has been the vacuum servo. A typical vacuum servo is described in GB-A-2009871, and the construction and operation of such servos will be well known to those skilled in the art. Whilst vacuum servos have been widely adopted within the motor industry they do have two significant defects. Firstly, they are of a substantial size and this gives rise to difficulties in locating the servo within the limited space available within the engine compartment of many modern cars. Secondly, they require a reliable source of vacuum for operation. The necessary vacuum can be obtained from the inlet manifold of normally aspirated vehicles. However, direct fuel injection engines (both diesel and petrol), turbocharged engines and super-charged engines cannot provide the necessary inlet manifold vacuum and accordingly in vehicles fitted with this type of engine it is necessary to provide a separate vacuum pump and vacuum reservoir if a conventional vacuum servo is to be used. This further complicates the installation problem

referred to above and in any event does not solve the basic installation problem caused by the large size of the vacuum servo itself.

With a view to overcoming the difficulties outlined above it has been proposed to provide a servo which is powered using hydraulic power from an accumulator which is charged by a hydraulic pump controlled by pressure switches. Whilst such a system reduces the size of the servo, the provision of the accumulator undesirably adds to the cost and weight of the braking installation. Further, since the accumulator is permanently charged to a high pressure the control valves utilized to control the application of hydraulic power from the accumulator to the servo are permanently subjected to a high pressure differential. This high pressure differential necessitates very accurate component construction and increases the risk of seal failure in use. An alternative proposal has been to provide the servo with a pump which delivers hydraulic fluid to a servo chamber of the servo when required. With such arrangements, when a desired pressure is reached in the servo chamber the output of the pump is connected to reservoir via a throttle opening whereby the desired pressure is maintained in the servo chamber and the excess output from the pump is delivered to the reservoir via the throttle. In such a system the circulation of actuating fluid from the reservoir through the pump and the throttle back to reservoir induces a temperature rise in the fluid during sustained braking and this temperature rise becomes unacceptable during prolonged periods of braking, for example whilst descending a mountain pass.

The present invention provides a servo for a hydraulic master cylinder in which a pump is provided for supplying the required pressurized working fluid to the servo chamber, but in which the problem of fluid heating is avoided.

According to one aspect of the present invention a servo for a hydraulic master cylinder comprises: a body defining a bore; a servo piston mounted in the bore for providing an output force to a piston of the master cylinder; a servo chamber defined between the bore and the servo piston for receiving pressurized fluid to drive the servo piston along the bore; a pump for supplying pressurized working fluid to the servo chamber; an input member for receiving an input force for an operating pedal; and control means operative when the pressure in the servo chamber reaches a value determined by the magnitude of the input force to relieve the pump load whilst maintaining the pressure in the servo chamber at the desired level.

In a particularly preferred embodiment of the invention the output of the pump branches with a first branch leading to the servo chamber via a non-return valve. The second branch is connectable to reservoir via the control means. When the brakes are applied, the control means is operative to close the second branch until the required pressure is generated in the servo chamber whereupon the second branch is opened to permit the output of the pump to be connected to reservoir whilst the pressure in the servo chamber is maintained by the non-return valve in the first branch. With such an arrangement, substantially no throttling of the pump output need occur when the second branch is connected to reservoir, and accordingly the pump can freely circulate working fluid without inducing any substantial temperature rise.

If a further increase in brake pressure is required the control means is operative again to close the second branch passage allowing the pump output to flow to the servo chamber via the non-return valve. If less braking force is required the control means is operative to connect the servo chamber to reservoir until the required

reduction in pressure in the servo chamber has been achieved.

In a particularly preferred embodiment of the invention the control means comprises a mechanical control valve which is operated by the input member. Conveniently, the control valve may be formed by a valve member slidably mounted within a bore provided in the servo piston and ports connecting the bore to the exterior surface of the servo piston. In this embodiment of the invention the pressure within the servo chamber is preferably communicated to the end face of the control valve member for generating a reaction force on the input member. Hence, the reaction force generated on the input member is directionally proportional to the pressure subsisting in the servo chamber.

Preferably, the servo chamber is connected to reservoir when the brakes are not in use and upon initial application of the brakes the control valve is effective firstly to isolate the servo chamber from reservoir and secondly to isolate the output of the pump from reservoir so that the pump output is directed to the servo chamber.

Whilst the control means may be a purely mechanical control valve arrangement, various advantageous embodiments of the invention utilize control means incorporating electrically operated valves. In one such embodiment of the invention operation of the pump and of the valves is under suitable electronic control to enable the brakes to be applied automatically for traction control purposes.

In a further and particularly preferred embodiment of the invention the magnitude of the input force is determined by a first electrical or electronic load sensing device and the magnitude of the pressure generated in the servo chamber is determined by a second electrical or electronic load sensing device. The outputs

of the load sensing devices are connected to an electronic control unit which is operative to control a valve to relieve the pump load when the pressure in the servo chamber reaches the level required by the input force. Such an arrangement is particularly desirable since the relationship between input force and pressure generated within the master cylinder (i.e. the "boost ratio" of the servo) can readily be varied under electronic control. Hence, if appropriate load or G-sensing devices are incorporated within a braking system the boost ratio of the servo can automatically be controlled to produce optimum braking conditions under driver only load or any other load condition.

The invention will be better understood and further features and advantages thereof will become apparent from the following description of three embodiments thereof, given by way of example only, reference being had to the accompanying drawings wherein:

Figure 1 illustrates schematically in longitudinal cross-section a first embodiment of the invention;

Figure 2 illustrates schematically in longitudinal cross-section a second embodiment of the invention; and

Figure 3 illustrates schematically in longitudinal cross-section a third embodiment of the invention.

Referring firstly to Figure 1 there is illustrated a servo 1 in accordance with the present invention. The servo in this case forms part of a powered tandem master cylinder 2. To this end, the servo piston 3 is formed integrally with the primary piston 4 of the master cylinder. As will be well understood by those skilled in the art the master cylinder 2 also includes a secondary piston 5 which separates the primary pressure

chamber 6 of the master cylinder from a secondary pressure chamber 7. The primary pressure chamber 6 is connected to one brake circuit via an output 8 whilst the secondary pressure chamber 7 is connected to another brake circuit by way of an output 9. In the rest condition illustrated in Figure 1 the pressure chambers 6,7 are connected to respective reservoir ports 10,11 by means of open centre valves 12. When the brakes are applied, the primary piston 4 moves to the left as viewed in Figure 1 allowing the centre valves 12 to close to separate the pressure chambers 6,7 from their associated reservoir ports and thereafter pressure is generated in the primary and secondary pressure chambers 6,7.

Referring now to the servo 1, the body 13 thereof is integral with the body of the master cylinder and is provided with a bore 14 coaxial with and of equal diameter to the bore of the master cylinder. The servo piston 3 is slidably mounted in the bore 14 and defines a servo chamber 15 to which pressurized working fluid from a pump 16 can be supplied via a passage 17. The passage 17 includes a first branch 17A which extends via a non-return valve 18 to the servo chamber 15 and a second branch 17B which extends from upstream of the non-return valve 18 to terminate in a port in the wall of the bore 14.

Preferably, the working fluid for the pump 16 is hydraulic brake fluid which is supplied from a suitable reservoir via a reservoir port 19, an annular chamber 20 and an inlet passage 21.

A control valve member 22 is located within a bore 23 formed in the servo piston 3. The valve member 22 is biased into engagement with an input member 24 by a light spring 25 located in a chamber 26 defined between the left-most extremity of the control member 22 and the end wall of the bore 23.

Radial bores in the servo piston connect the

servo chamber 15 to an annular chamber 27 which is in turn connected to a longitudinal passage 28 within the control valve member by radial bores provided in the control valve member. Hence, the servo chamber 15 is in permanent communication with the chamber 26.

Radial passages 29,30 connect the bore 23 to the annular chamber 20 and a radial passage 31 connects the bore 23 to an annular chamber 32.

When the brakes are not in use, the various components are in the illustrated positions and the pump 16 is not energized.

In use, means are provided to detect initial pressure on or movement of the brake pedal, and in response to such pressure or movement the pump 16 is energized. In the initial configuration of the components the pump output flows via passage 17 and the second branch passage 17B to the annular chamber 32 and from there via passage 31, annular chamber 33, bore 30 and annular chamber 20 to the inlet passage 21. Hence, fluid can circulate freely and no pressure is generated within the servo chamber 15.

As the brake pedal is depressed the input rod 24 is moved to the left as viewed in Figure 1 and causes corresponding left-ward movement of the control member 22. This movement firstly causes the land 34 on the control member 22 to close the passage 29 thereby isolating the chambers 15 and 26 from reservoir. Next, the land 35 of the control member 22 closes the passage 31 thereby isolating the pump output from the pump input and from reservoir. Accordingly, the pump output is forced to flow via non-return valve 18 and first branch passage 17A to the servo chamber 15 where it produces a pressure increase to move the servo piston 3 to the left and initiate pressure generation within the primary and secondary chambers 6,7. Left-ward movement of the servo piston is accompanied by corresponding movement of input member 22 and the brake

pedal accordingly descends in conventional manner as the brakes are applied.

As pressure within the servo chamber 15 increases, pressure within the chamber 26 increases to produce a reaction force on the control member 22 which is transmitted to the input member 24 and thence to the brake pedal to provide appropriate "feel" to the brake pedal.

When the desired brake pressure has been generated the driver will cease the downward movement of the brake pedal and if he wishes to maintain the present degree of braking he will hold the pedal at a fixed position. This will in turn hold the control member 22 at a fixed position relative to the body 13. Pump 16 will continue to supply pressurized fluid to the chamber 15 and accordingly there will be a slight further forward movement of the servo piston 3. This slight forward movement will be sufficient to move the land 35 clear of the passage 31 and accordingly a non-restricted circulation passage will be established from the pump outlet passage 17 via the branch 17B, annular chamber 32, passage 31, annular chamber 33, passage 30, annular chamber 20 and passage 21. Forward movement of the servo piston 3 will be insufficient to open passage 29 and accordingly no connection between the servo chamber 15 and the reservoir will be established. The non-return valve 18 in branch passage 17A will prevent the return flow of fluid through the passage 17A. Thus, the desired level of pressure within the servo chamber 15 will be maintained whilst the pump 16 is able freely to circulate fluid and accordingly there will be no excessive heat generation.

If the driver wishes to reduce the level of braking produced he will lift he foot slightly on the pedal and this will cause rearward movement of the control member 22 relative to the servo piston 3 sufficient to uncover the port 29 and accordingly allow fluid to flow from the servo

chamber 15 to the reservoir. If the brakes are only partially released a new equilibrium condition will be reached corresponding to the force applied by the driver to the pedal. If the pedal is completely released the servo chamber 15 will be able to vent completely to the reservoir via the passage 29 and the components will return to the illustrated configuration. The pump will be stopped.

The above described arrangement offers a number of significant advantages. Firstly, as compared with servos operating from a hydraulic accumulator, there is no requirement for any seal to be subject to a permanent pressure differential. In the rest condition of the components no part of the system is pressurized and accordingly the pressure seals are required only to withstand pressure differentials during actual braking. The pump 16 is only required to deliver pressure up to the level determined by the driver during any particular braking operation. In other words, the output pressure of the pump never exceeds the pressure for the time being in the servo chamber 15. When the desired pressure within the servo chamber 15 is reached the pump is automatically unloaded and can circulate fluid freely. This combination of features substantially reduces the duty requirements of the pump and would enable, for example, an existing ABS motor to be used to drive the pump.

It will be appreciated that if the servo system fails the master cylinder will operate as a conventional unassisted tandem master cylinder and there will only be a small loss of pedal travel corresponding to the movement necessary to bring the control member 22 into engagement with the end face of the bore 23. If one of the braking circuits fails, the servo will continue to provide servo assistance to the other braking circuit. By utilizing brake fluid as the working fluid for the servo system the entire interior of the body 13 is filled with brake fluid

and complicated arrangements to separate the fluid of the servo from the fluid of the braking system are not required.

Referring now to Figure 2, the powered master cylinder 102 is substantially identical to that described above with reference to Figure 1 save that the reservoir port 19 of the embodiment of Figure 1 has been omitted. An additional non-return valve 103 is provided in the outlet passage 104 of the pump 116 and a by-pass passage 105 is provided to connect the outlet passage 104 to the inlet passage 106. A first electrically controlled solenoid valve 107 is provided in the inlet passage 106 and a second electrically controlled solenoid valve 108 is provided in the by-pass passage 105. A reservoir connection 109 is connected to the inlet passage 106 and to the by-pass passage 105.

In the rest condition the various components are in the illustrated configuration - that is valves 107 and 108 are normally open. If the brakes are applied by depressing the foot pedal the pump 116 is energized and the valve 108 is closed. The system then operates substantially as described above with reference to Figure 1.

In addition to this normal operation the embodiment described in Figure 2 is adapted to provide automatic brake application for traction control purposes. If wheel spin during power application is detected both solenoid valves 107 and 108 are automatically closed and the pump 116 is energized. In consequence, pressure is built up in the servo chamber 110 and both braking circuits are pressurized. The ABS modulator of the braking system then isolates the non-spinning wheel from the effects of the applied brake pressure and accordingly the brake pressure has the effect of braking the spinning wheel only. When sufficient pressure is applied to the spinning wheel

to bring the spin under control the solenoid valve 108 is opened to unload the pump. The brake pressure within the braking system is then maintained by non-return valve 103 until further changes are necessary as sensed by the ABS/traction control monitors. If further pressure must be applied then the valve 108 is again closed and if pressure must be relieved both solenoids 107 and 108 are moved to their open position.

A further modified embodiment of the invention is illustrated in Figure 3. In this embodiment the servo piston 201 is slidably mounted within the bore 202 of a tandem powered master cylinder. A servo chamber 203 is defined between the piston 201 and the bore 202. Pressurized working fluid may be supplied to the servo chamber 203 from a pump 204 via a branched supply passage 205. The second branch 205B of the supply passage connects the servo chamber 203 to the pump 204 via an electronically controlled proportional solenoid valve 206. The first branch 205A of the passage connects the servo chamber 203 to the pump output via a non-return valve 207. A reservoir connection 208 on the master cylinder/servo body 209 provides a connection between a fluid reservoir and an annular chamber 210 defined about the periphery of the servo piston 201. A passage 211 extends from the annular chamber 210 to the pump input 212 and to a reservoir chamber 213 defined within the valve 206.

An input member 214 which is connected to the brake pedal is connected to a thrust member 215 which is mounted within a bore defined in the end of the servo piston 201. An input load cell 216 is located between the thrust member 215 and the end surface of the servo piston 201 to measure the input load applied by the input member 214 to the servo piston 201. The output of the load cell 216 is connected to an electronic control unit 217 for controlling the pump 204 and the valve 206. An output load

cell 218 is provided within the master cylinder body for measuring the output pressure of the master cylinder. In the illustrated embodiment, the output load cell 218 is loaded by a piston 219 fitted in the end of the master cylinder bore. However, other arrangements are possible. The output of the load cell 218 is also connected to the electronic control unit 217.

In the illustrated rest condition of the components the servo chamber 203 is connected to reservoir via the branch passage 205B, a passage 220 formed in the spool 221 of the valve 206, the chamber 213, passage 211, chamber 210 and reservoir port 208.

In use, when the brakes are applied the input member 214 applies a force to the servo piston 201 and this force is detected by the load cell 216. Upon detection of this force the control unit 217 energizes the pump 204 and moves the spool 221 upwardly which firstly closes port 222 to isolate the servo chamber 203 from reservoir and then closes port 223 to isolate the pump output from reservoir. Thereafter, the pump output flows via branch passage 205A and non-return valve 207 to the servo chamber to energize the master cylinder substantially as described above with reference to Figure 1.

As pressure within the brake system rises this pressure will be detected by load cell 218 and a corresponding signal supplied to the control unit 217. When the brake system pressure rises to a level determined by the applied input force the control unit will move the spool 221 of the valve 206 downwardly to open the port 223 and relieve the pump output to reservoir via the passage 220, chamber 213, passage 211, etc. Downward movement of the spool will, however, be insufficient to open port 222 and accordingly the desired pressure will be maintained in the servo chamber 203. When less braking force is required the pedal is allowed to rise and the force applied to the

servo piston 201 by the input member 215 is reduced. The system will under these circumstances detect that the pressure within the braking system is too high relative to the force measured by the load cell 216 and the control unit 217 will lower the spool 221 further to open port 222 and allow pressure to be relieved from the servo chamber 203. Accordingly, during varied braking conditions movement of the servo piston 221 is controlled by the control unit in response to the outputs of the load cells 216 and 218 to maintain the correct level of system braking pressure.

A particular advantage of the arrangement of Figure 3 is that the "boost ratio" of the servo can be varied by the control unit 217. In other words, the amount of brake pressure generated by a particular input force can be varied by varying the reaction of the valve spool 221 to the detected brake system pressure for any particular input force. This offers a number of advantages. For example, if the vehicle is fitted with air suspension or an axle load measuring device the boost ratio can be increased when the vehicle is laden so that a substantially constant pedal pressure is required to achieve a particular level of deceleration regardless of vehicle load. Also, if the braking system is fitted to an electric vehicle with regenerative braking the boost ratio can be varied to take account of the power absorbed by the regenerative system when the brakes are applied. If the vehicle to which the system is fitted incorporates a cruise control system having vehicle distance and sensing arrangements the servo can be used to automatically apply the brakes to maintain the correct time interval between lead and following vehicles. As in the embodiments of Figures 1 and 2, if the servo system fails the master cylinder acts as a simple non-powered tandem master cylinder with substantially no loss of pedal travel. In addition to controlling operation

of the pump 204 and spool valve 206 the control unit 217 may be utilized to actuate brake lights in response to a detected input force from load cell 216 or the compare the actual deceleration achieved with the pressure generated within the braking system so as to indicate brake fade or severe overloading of the vehicle.

Although in the illustrated embodiments of the invention the diameter of the bore in which the servo piston moves is equal to the diameter of the bores of the primary and secondary chambers of the master cylinder this need not be the case, and if desired the servo bore may be made larger in order to increase the boost ratio of the system.

By appropriate design it is accordingly possible to produce a system which will provide a brake output pressure of the highest possible value required without loss of servo effect. This is in contrast to convention vacuum servos which, because of the large piston area necessary to generate high pressures, can in some designs only offer limited servo assistance to the driver.

Although the pump motor associated with the servo motor may be run at a constant speed it may be desirable to control the motor e.g. by pulse width modulation of the power supply. Such an arrangement may be desirable to prevent the motor from speeding up when the desired brake pressure is reached and the motor is unloaded. It may also be useful as part of an ABS braking system in which pulse width modulation of the servo pump may be useful to improve pedal feel during ABS operation by slowing down the rate of perceived vibration of the brake pedal. In this connection it will be appreciated that because the servo pump is relieved whenever the input force is held at a constant value, the motor used to drive the pump can, except when required to drive the servo pump to increase brake pressure, be used for other purposes, for example to drive the pump of an ABS system.

11. A servo according to claim 10, wherein operation of the pump and of the valves is under suitable electronic control to enable the brakes to be applied automatically for traction control purposes.

12. A servo according to any preceding claim, wherein the magnitude of the input force is determined by a first electrical or electronic load sensing device and the magnitude of the pressure generated in the servo chamber is determined by a second electrical or electronic load sensing device.

13. A servo according to claim 12, wherein the outputs of the load sensing devices are connected to an electronic control unit which is operative to control a valve to relieve the pump load when the pressure in the servo chamber reaches the level required by the input force.

14. A servo for a hydraulic master cylinder substantially as hereinbefore described with reference to the accompanying drawings.

Patents Act 1977
Examiner's report to the Comptroller under Section 17 (The Search Report)

Application number

GB 9211759.7

Relevant Technical fields

(i) UK Cl (Edition L) F2F (FFB, FFD)

(ii) Int Cl (Edition 5) B60T

Databases (see over)

(i) UK Patent Office

(ii) ONLINE DATABASES: WPI

Search Examiner

PETER SQUIRE

Date of Search

29 JUNE 1993

Documents considered relevant following a search in respect of claims 1-14

Category (see over)	Identity of document and relevant passages	Relevant to claim(s)
	NONE	

Category	Identity of document and relevant passages	Relevant to claim(s)

Categories of documents

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